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ATIGUE RESISTANT STRUCTURES

C R SMITH

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FATIGUE RESISTANT STRUCTURES

by

C. R. Smith

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62 p

Engineering Department

CONVAIR

A Division of General Dynamics Corporation

(San Diego)

CONTENTS

ILLUSTRATIONS

SUMMARY:

The work for the fiscal year of 1958 on Basic Fatigue Research has been directed towards obtaining methods for designing fatigue resistant structures. Two such methods have been obtained: (1) the use of rivets driven through the edge of splice doublers, and (2) the use of thin auxiliary doublers to permit using extra rivets away from the high stress area in the main splice doubler.

Test data show that a substantially lighter structure could be had for the same fatigue life by using either of the two methods, or a lifetime of up to twenty times that of an equivalent weight structure of conventional design.

Thin doublers are being used in the Models 880 and 600. Edge driven rivets have been approved for operators of commercial airlines in repairs or as fatigue inhibitors of airplanes now in service. Convair has a patent pending on edge driven rivets.

Miscellaneous data are also presented on fatigue behavior and on photoelastic analysis of stress distribution in simulated and built up structures.

(B) INTRODUCTION:

Research on fatigue is generally comprised of one or more of the following: 1) basic nature of fatigue on the atomic structure level, 2) methods of estimating fatigue life, and 3) improving design of fabricated structures.

Other than the use of shot peening for improving fatigue life, and reducing stress concentrations, very little to date has been published on improving fatigue life of fabricated structures. Most of the published works have had to do with either item 1, or 2.

The major portion of research at Convair has been directed toward the third item, improving basic design; although some of the earlier work was on methods of estimating life. [1,2] This was brought about by the immediate need for designing fatigue resistance in our airplanes.

The author wishes to express his appreciation to Mr. G. D. Lindeneau and other members of the Structures Laboratory for valuable assistance in obtaining test data in this research.

THEORY:

Background

It is well known that metals when subjected to repeated loading will eventually fail in fatigue at stresses far below the ultimate or even yield strength of the material. The relationship between these stresses and longevity are usually plotted in the form of Stress versus Number of Cycles curves or S-N curves. Whether the fatigue loading is repeated, reversed, or somewhere between, can be identified by the stress ratio, symbolized as R, or the ratio of the minimum stress divided by the maximum stress. A family of curves[3] for smooth axially loaded 7075-T6 round bar specimens is given in Figure 1. According to the notation above, the curve for R= -1 is for completely reversed loading, and for where R = 0 is for repeated loading.

Fabricated structures behave very much like smooth specimens, with the exception that stress concentrations afforded by discontinuities may cause actual stress to be many times nominal values. Were we able to determine actual stresses at concentrations, the problem in estimating fatigue life would be infinitely simpler. The next best thing is to understand relative stresses at and near the concentrations. Sometimes localized stresses at concentration points cause yielding which result in re-distribution of stress. If re-distribution can be carried far enough, that is to say, if distribution could be carried to the extreme so that the effect of stress concentrations could be completely eliminated, fatigue would not be a problem.

THEORY (Contd)

A survey of service failures in airplanes discloses that over 75% of the failures involve rivets. Furthermore, these are particular rivets, being the first ones engaged in a splice or other attachment. Needless to say, if the fatigue problem were solved in just this one location, we would be well on our way to fatigue free airplanes.

The first step in this analysis is to find out how the rivet behaves in this particular location. It is obvious that there is too much stress here and that some readjustment in load distribution would be in order. An obvious solution would be to design such as to make this particular rivet or row of rivets incapable of carrying so much load.

Before studying rivets, however, some more elementary concepts are in order.

Elementary Concepts

Take the case of a single hole in an infinitely wide sheet. Classic theory and experimental observations have shown that the stress in a loaded sheet at the edge of the hole is roughly three times that in the sheet away from the hole. This distribution is shown in Figure 2,[4] and holds so long as materials remain in the elastic range. However, on reaching the yield point, a re-distribution of stress takes place.

In the case of fatigue, the stress at the edge of the hole is all we need consider. If we know actual values of stress, ordinary S-N curves for smooth specimens are all we need to predict fatigue life.

THEORY (Contd)

Linear Strain Theory

The theory of linear strain was first presented before the Society for Experimental Stress Analysis in a paper entitled "Prediction of Fatigue Failures in Aluminum Alloy Structures" by C. R. Smith[1] in April, 1954. Since then, essentially the same theory was independently developed by Gunn[5] of England. In essence, it states that a concentration is a geometric relationship and would be more appropriately expressed in terms of strain rather than stress. Also, this relationship should hold irrespective of whether the material is in a state of elastic or plastic deformation.

Accordingly, the stress distribution shown for the hole specimen would be interchangeable with strain distribution in either the elastic or plastic range. Thus, knowing strain values, it is an easy matter to find the corresponding stress from ordinary stress-strain curves. Thus, were the nominal stress away from the hole 35,000 psi and the material were 7075-T6, the corresponding nominal strain would be .0033 in./in. (assuming E = 10,500,000 psi).

Assuming that the stress concentration factor of 3 applied to strain, we can multiply .0033 x 3 and get .0099 in./in. peak strain. According to the stress-strain curve shown in Figure 3,[5] the corresponding stress would be 77,000 psi. On relieving load, however, the stress-strain relationship would fall along line AC. Whereas the stress-strain graph (AB) for unloading in a smooth specimen would stop at point B, because of the relatively large amount of material still in the elastic range (only a

THEORY (Contd)

very small amount of the total cross section suffered plastic deformation), the graph will continue on to a point C where equilibrium between the plastically deformed and elastically deformed material is reached.

For fatigue testing at this load level, then, the cyclic stress would fall between points C and A, and the fatigue life could be predicted, using ordinary S-N curves as shown in Figure [illegible]. Furthermore, subsequent cyclic loading at lower stress levels, instead of originating at point O would start at point C, resulting in a much longer life than a similar structure with no pre-stress history. This stems from the fact that the peak stress is lowered by an equivalent amount of the residual stress. Thus, were the low level amplitude before pre-stress as shown in Figure 4a, and high level as shown in Figure 4b, subsequent low level loading would be as in Figure 4c.

Sample S-N curves for coupons having centrally drilled holes for a theoretical concentration factor of 2.4 (based on net area) with and without pre-stress are given in Figure 5. Note that the calculated and experimental values are in good agreement where pre-stressing was present. Without pre-stress, however, the calculated values were somewhat short of the experimental data. This is attributed to unknown amounts of beneficial residual stresses set up by drilling which were not considered in the calculations. Experimental observations using PhotoStress Plastic[6] confirm this. These are described under Experiment B of this report.

Accordingly, in our experimental S-N curves shown in Figure 5, we can assume that the pre-stress was sufficient to wipe out the effects or residual stresses due to drilling.

THEORY (Contd)

In the case of riveted joints, a reduction of fatigue life is experienced by small amounts of pre-stressing. This is apparently caused by loss of propping action normally afforded by the rivets. This is illustrated in Figure 6. In Figure 6a is shown an assumed cyclic amplitude for repeated loading on a one-inch wide lug specimen loaded by quarter inch diameter bolt. In Figure 6b is shown the same specimen, but this time a quarter inch diameter rivet is used in place of the bolt. Assuming that the driving force of the rivet is sufficient to swell the hole .00075 inch in diameter, the peripheral tension set up around the hole would amount to .003 in./in. or an equivalent of 30,000 psi stress with no external load applied.

When external load is applied, the strain rate at the edge of the hole is no longer the same as for the bolt loaded lug, but will be in accordance with a new spring rate, including the stiffness of the rivet. Accordingly, the same load that caused the stress to cycle from zero to maximum in Figure 6a, will cause only partial stress cycles as shown in Figure 6b, the lower value depending on the amount of rivet driving force and relative stiffness.

The upper part of the cycle is usually unaffected by the rivet driving force because of the limited compressive deformation stored up in the rivet. For example, where we assumed that the diameter of the hole to be increased .00075 in., a certain amount of stored up compressive deformation would have to be absorbed in the rivet. Again assuming that the rivet to be three

THEORY (Contd)

times* as stiff as the surrounding material, a compressive deformation of .00025 in. remains stored up in the rivet, and the added stiffness of the rivet would disappear after an additional .00025 deformation (in addition to the .00075) occurred. This total of .001 in. deformatic would amount to a strain of .004 in./in., since the hole was 1/4 in. di or an equivalent stress of 40,000 psi if the lug were aluminum alloy. A ingly the peak stress in Figure 6b would be higher than that shown in 6a if it fell within the 40,000 psi range for the particular set of values in this example.

The assumed .00025 compressive deformation in the rivet can be like to "spring back" commonly experienced in forming operations. The fact t this compressive deformation decreases the tensile strain rate at the edge of the hole is somewhat confusing. It can be shown experimentally, see Figure 7, that the spring constant for two springs opposing each oth is exactly the same as when they act together. Accordingly, the result added stiffness.

A numerical example of the above will illustrate these particular effects on fatigue life.

Let's assume that this lug is subjected to repeated loads producing a nominal net stress of 12,500 psi. According to the geometric configura tion, Ref. Figure 8, the stress concentration factor would be about 4.8. Accordingly, the maximum stress would be 4.8 x 12,500 or 60,000 psi. The fatigue life, according to the graphs shown in Figure 1, would be 45,000 cycles.

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* This approximation is born out by PhotoStress studies, see Experiment B in this report.

THEORY (Contd)

With the tightly driven rivet, the cyclic stress would not be 0-60,000 psi, but from 30,000 to 60,000 psi (assuming that the rivet expands the hole .00025 in.). The corresponding life for this would be 2,500,000 cycles. If this same specimen were overloaded once statically to cause elongation of the hole sufficient to relieve the residual tension stress, a permanent set of .004 in./in. would be required. In other words, the specimen would have to take on a residual compressive stress of 40,000 psi (corresponding to strains of .003 in./in. stored up in the specimen itself, plus .001 in./in. for the rivet) to comply with these dimensional changes. The subsequent fatigue life for the same cyclic loading would be 1,000,000 cycles.

It is of interest to note that this trend seems to refute data obtained by other researchers, [8] where static preloading generally produces an increase in fatigue life. Suffice to say that these data are not directly comparable in that riveted joints usually contain more than one row of rivets, so that prestressing, if high enough, will re-distribute load between rivets. This generally more than compensates for loss of propping action afforded by the rivet. However, the trend will be for a reduction of fatigue life for small amounts of preloading, [9] and by an increase in life where large preloads are present. This will be covered more fully in the experimental portion of this report.

EXPERIMENTS:

A. VINYL PLASTIC MODELS FOR STRESS DISTRIBUTION BETWEEN RIVETS

In Figure 9 is shown a riveted joint simulated by fastening a one-inch wide strip of 0.10 thick vinyl plastic to an 0.020 thick piece of cellulose-acetate with three screw paper fasteners. The model is sandwiched between two polaroid filters so that when light is passed through the assembly, interference fringes are set up due to load in the vinyl.*

This is called photoelasticity or principle of birefringence which is described in Ref. 4. Briefly, however, any birefringent material (material having photoelastic properties) when stressed will cause light to travel faster in the direction of tension and slower in compression.

Light, on passing through the first polaroid, is constrained to vibrate in a single plane. Provided that no stress exists in the plastic model, it will emerge and pass through the second polaroid unchanged. However, if the plastic is loaded, the beam of light is split into two beams (double refraction) whose wave motions are at right angles to each other. The beam whose vibration is

* The technique of using vinyl plastic was first described by Dr. M. Hetenyi of Northwestern Univ. at the Spring Meeting of the Society for Experimental Stress Analysis, Indianapolis, May, 1952. Papers for the seminar in which this was disclosed were never published.

EXPERIMENTS: (Contd)

in the direction of tension will emerge ahead of the other, so that when re-combined by the second polaroid, interference between the two beams will set up color fringes which can be calibrated in terms of differences in strains. If the polaroids are oriented so that their axis are in, or normal to, the direction of tension, the differences will be of maximum principal strains.

These fringes, however, appear as dark lines in the photograph and amount to about .050 in./in. strain per fringe. The second order fringe appeared at the first rivet before any color was observed in the center rivet. Stress patterns appear only in the vinyl half of the joint because double refraction does not occur in the cellulose-acetate.

The above is a visual demonstration of the difference in load between rivets. The fact that this was a plastic model does not in the least detract from the similarity to a real structure. This also shows the reason why fatigue failures occur along the first line of rivets in an airplane structure and why more effort needs to be made in reducing the differences in stresses between rows of rivets.

EXPERIMENTS: (Contd)

B. PHOTO-STRESS STUDY OF STRESSES AROUND HOLES AND RIVETS

In Figure 10 is shown a sample of 0.10 thick 2024-T3 aluminum alloy sheet covered with PhotoStress(7) plastic. The plastic coating was applied prior to drilling and spotfaced at the rivet location to permit forming the head. The dark areas around the holes and rivet represent residual stresses set up by drilling and rivet forming. The hole at the extreme left was drilled with a sharp drill, and that in the center with a dull drill. The rivet was countersunk on the far side.

PhotoStress is a trade name for a special birefringent plastic that can be cemented to an actual part for viewing stresses. The particular type used in this experiment was .048 in. thick. One fringe order represents about 14,000 psi stress in aluminum alloy.

A similar specimen was prepared with open hole, and rivet filled holes prior to application of the PhotoStress. The rivet heads were milled flush to permit applying the PhotoStress. This specimen was loaded axially and observations of strains adjacent to the edge of the hole, rivet, and away from points of concentration. These data are presented in Figure 11. Note that the slope of the graph for the rivet filled hole is almost as steep as that for the specimen away from the hole.

EXPERIMENTS: (Contd)

In the case of the hole filled with a stainless steel rivet, the slope is slightly steeper than for the specimen away from the hole. However, apparently because of the lack of spring back in the steel, a greatly reduced slope occurred after application of about 24,000 psi nominal gross stress.

It is to be noted that the strain values given here are the differences in principal strains. At the edges of the holes and rivets, it can be assumed that the normal strain is zero and that the values given are e_{max}. However, in the case of the control, some correction would be required to give the strain in the direction of load. This would amount to about 12% less strain than shown to conform with a rated value of $E = 10.3 \times 10^6$ for 7075-T6.

From the above, the following conclusions may be drawn:

1. Aluminum alloy structures with steel rivets should have a longer fatigue life than with aluminum alloy rivets.

2. The strain at a concentration is linear in the plastic range of stress. Plastic flow* should have started in this alloy at about .007 in./in. strain, yet the slopes of all curves are linear even to .018 in./in. strain,

* The plastic became unbonded during strain readings at the 72,000 psi stress level, so that values for the control and permanent set were unobtained.

EXPERIMENTS: (Contd)

C. EFFECT OF RIVET DRIVING FORCE

From the preceding experiment one can assume that a superior fatigue life could be had by driving the rivets harder.

In this experiment four sets of 0.10 clad 7075-T6 by one inch wide lap joints were tested at 10,000 psi nominal gross cyclic stress, R = 0.1. They were fastened with three 1/4" diameter 100° countersunk 2024-T4 rivets with one-inch spacing between rivets and 1/2" edge distance. On the assumption that the swelling force of the rivets was a function of the diameter of the formed head, five specimens each with head diameters of .315, .330, .345 and .360 inches were tested. The .330 or 1-1/3 diameter head was considered to be a control.

These data are plotted in Figure 12. Note that those specimens having nearly 1-1/2 diameter heads (.360) had nearly 50% greater life than those with 1-1/3 diameter heads.

D. BENDING STRESSES IN A RIVETED JOINT LOADED IN TENSION

Figure 13 shows measurements of nominal stress versus strain adjacent to the first rivet in a two rivet lap joint. The joint was made of 7075-T6 material having an upper sheet of 0.10 x 1-1/2 and lower sheet of 0.125 x 1 with two 3/16 diameter rivets spaced one inch apart. Strain measurements were made with Baldwin SR-4 resistance type strain gages which were mounted back to back on the

EXPERIMENTS: (Contd)

1/4 in. wide flange alongside the first rivet in the 0.10 material.* Note that the eccentricity between sheets causes the strain on one side to be roughly twice that of the average.

Figure 14 shows similar measurements for the doubler of a butt splice having two rivets on each side of the butt. Here the difference between the strains on the high side and nominal values is even more exaggerated.

The above experiments show that bending stresses in single shear joint construction constitutes one-half or more of the total stress. Accordingly, a substantial improvement in fatigue strength could be had if bending could be eliminated at the first loaded rivet. Also it would appear that, since that portion of stress due to bending in the doubler plate is higher than the axial stress, an improvement in fatigue life of the doubler could be had by making the doubler thinner. Data for these are included in Experiment E.

E. FATIGUE TESTS ON LAP WITH EDGE DRIVEN RIVETS

Figure 15 shows an edge view of simple two rivet lap joints, except that Figure 15b shows the addition of two extra rivets to relieve bending stress. The fatigue life for the conventional

* Loading was done in a Baldwin 120,000 lb. hydraulic testing machine.

EXPERIMENTS: (Contd)

splice was 55,000 cycles,* while the life for that with the edge driven rivets was 248,000 cycles. The values given represent averages of four specimens each at repeated gross stresses of 10,000 psi.

In Figure 16 are given S-N curves of lap joints with and without edge driven rivets.(10) The specimens for these were made of 1 x 1 x 0.1 7075-T6 angle nested to permit engaging four 3/16 diameter rivets spaced one inch apart in tandem.

The angle was used here because it was felt that the sheet specimens as given in Figure 15, exaggerated the bending, and that nested angles would simulate the fixity afforded by conventional airplane structure using stringers.

All of the fatigue failures for these tests were in the sheet through the edge driven rivets, indicating that bending was still critical.

Examination of a similar axially loaded specimen with Photo-Stress showed that the maximum strain at the edge of the first rivet in the conventional splice was .007 in./in. while a similar location for the edge driven rivet gave .0028 in./in. for a nominal strain of .00133 in./in.

* All fatigue tests in this project were made on SF 1U and SF 20U Sonntag Fatigue Testing Machine.

EXPERIMENTS: (Contd)

While these fatigue tests do not show the improvement that could be expected from the PhotoStress survey, they are substantial, and indicate that a considerably lighter structure could be made for the same life using edge driven rivets.

F. THIN DOUBLERS (11)

An alternative to the idea of forcing bending stresses to take place at a rivet incapable of carrying load, would be to allow bending to take place, but reduce the axial load at the first rivet. This is done by inserting a thin doubler in between the faying surfaces and extending beyond the main splice area as shown in Figure 17b. The idea is to have the auxiliary doubler so thin that is it incapable of inducing a bending stress in the main sheet. Yet, the doubler has to be strong enough to take out a large portion of the axial load before entering the main splice area.

A comparison of fatigue strengths for conventional splices and those with thin doublers is given in Figure 18 . Further details are included in Ref. 11.

From the above, it would appear that thinner doublers for butt splices would be superior even without the auxiliary doublers. In Figure 19 are plotted data for fatigue lives of butt splices with doublers one gage thicker than the skin, equal to skin thickness, one gage thicker than the skin, and two gages

EXPERIMENTS: (Contd)

thinner than the skin.

It is to be noted that the poorest showing was made by those splices having doublers thicker than the skin; also that thicker materials had shorter fatigue lives.

G. EFFECT OF PRESTRESSING ON RIVETED LAP JOINTS

The object of this experiment was to evaluate the effect of prestressing on riveted lap joints. In Figure 20 are S-N curves for lap joints of .051 clad 7075-T6 without prestress, with single static prestresses of 18,000 psi, 25,000 psi, and 42,000 psi. Points plotted represent average values for four specimens each. Note that for 18,000 psi prestresses, the S-N curve falls below that of virgin specimens. However, for 25,000 psi prestress, there is very little change, and for prestress of 42,000 psi, the curve is substantially above that for virgin specimens.

DISCUSSION:

Despite the substantial increase in fatigue life obtained in these experiments, it should be emphasized that they are by no means optimum.

Further research is needed to take advantage of the weight saving afforded by higher strength alloys. While the state of the art is designing fatigue resistant structures may never reach the situation where 7075 or any other high strength alloy could be used to its fullest static strength, a lighter structure than presently employed could be had without sacrifice of fatigue life.

Of particular importance is the fact that the test programs carried on so far are more indicative of the direction in which research should be aimed than of solution to the problems. For example, we still do not have production design allowables in fatigue for various sheet thickness and rivet combinations. The trend seems to be for a lower allowable for greater thickness sheets. Accordingly, increasing the thickness of a sheet specifically for fatigue strength could result in very little improvement at a great expense in weight. This appears to be a result of increased eccentricities caused by the extra thickness, notwithstanding the fact that the moment increases as the thickness to the first power, while the section modulus increases as the square of the thickness. This would suggest that the lesser thicknesses are more able to reduce the effects of bending by re-alignment.

DISCUSSION: (Contd)

Whereas we now use failsafe design practices in all of our airplanes, there is a definite need for information leading to more reliable estimates on longevity. The linear strain theory of cumulative damage appears to be a simple method of estimating life from ordinary S-N data, however, more data for service type loading is needed to substantiate the method. More work along this line is planned on receipt of a program loading fatigue testing machine now on order. Also, it appears that large numbers of cycles at very low stress levels may improve fatigue life. This is commonly called "coaxing" and has been found to be true in steels, although aluminum alloys have been reported to be unaffected by "coaxing". A few tests run at Convair seem to indicate that aluminum alloy can be "coaxed", however, more work needs to be done before this can be established

CONCLUSIONS:

A. The majority of service fatigue failures in airframes are caused by 1) too much load carried by the first rivet or row of rivets, and 2) by too much bending at the first rivet or row of rivets in a joint.

B. A substantial increase in fatigue life can be had by thinning the doublers at the first row of rivets. This reduces both load at the rivet, and, bending stress in the sheet. A general rule for splices in aluminum alloy sheet would be to make the doubler at the first rivet in the splice not over one-half of the sheet thickness, if aluminum alloy doublers are used, or approximately 1/5 of the thickness if steel is used. Use as many extra rivets as required to make up for lack of static strength at these locations.

C. Rivets should be tightly driven. Experiments have shown that 50% longer life can be had by forming heads 1-1/2 times the shank diameter instead of the 1-1/3 diameter head commonly used.

D. Fatigue failures can be reduced by causing bending stresses to take place away from points of maximum axial stress. Rivets driven through the edge of doublers are very effective in doing this.

REFERENCES

1. C. R. Smith, "Prediction of Fatigue Failures in Aluminum Alloy Structures", Proceedings of the Society for Experimental Stress Analysis, Vol. XII No. 2, 1955, Cambridge 39, Mass.

2. C. R. Smith, "Fatigue--Service Life Prediction Based on Tests at Constant Stress Levels", Proceedings of the Society for Experimental Stress Analysis, Vol. XVI, No. 1, 1958.

3. F. M. Howell and J. L. Miller, "Axial-Stress Fatigue Strengths of Several Structural Aluminum Alloys", Proceedings of the American Society for Testing Materials, 1958, Table II.

4. Frocht, M. M., "Photoelasticity", Vol. 1 (1941), John Wiley & Sons, Inc., New York.

5. K. Gunn, "Effect of Yielding on the Fatigue Properties of Test Pieces Containing Stress Concentrations", The Aeronautical Quarterly, November, 1955.

6. ANC-5, "Strength of Metal Aircraft Elements", 1955, U.S. Govt. Printing Office.

7. F. Zandman & Marc R. Wood, "PhotoStress, A New Technique for Photoelastic Stress Analysis for Observing and Measuring Surface Strains on Actual Structures and Parts", Product Engineering, September, 1956.

8. A. O. Payne, "An Investigation Into the Fatigue Characteristics of a Typical 24S-T Aluminum Alloy Wing", American Society of Mechanical Engineers, Preprint, Session 8, Paper 3, November, 1956.

9. C. R. Smith, "The Fatigue Strength of Riveted Joints", Aircraft Engineering, February, 1957.

10. C. R. Smith, "Aircraft Fatigue - - Is Low Strength Material the Answer?", Preprint No. 780, 26th Annual Meeting of the I.A.S.

11. C. R. Smith, "Thin Doublers for Fatigue-Resistant Aluminum Alloy Structures", Aero/Space Engineering, February, 1959.

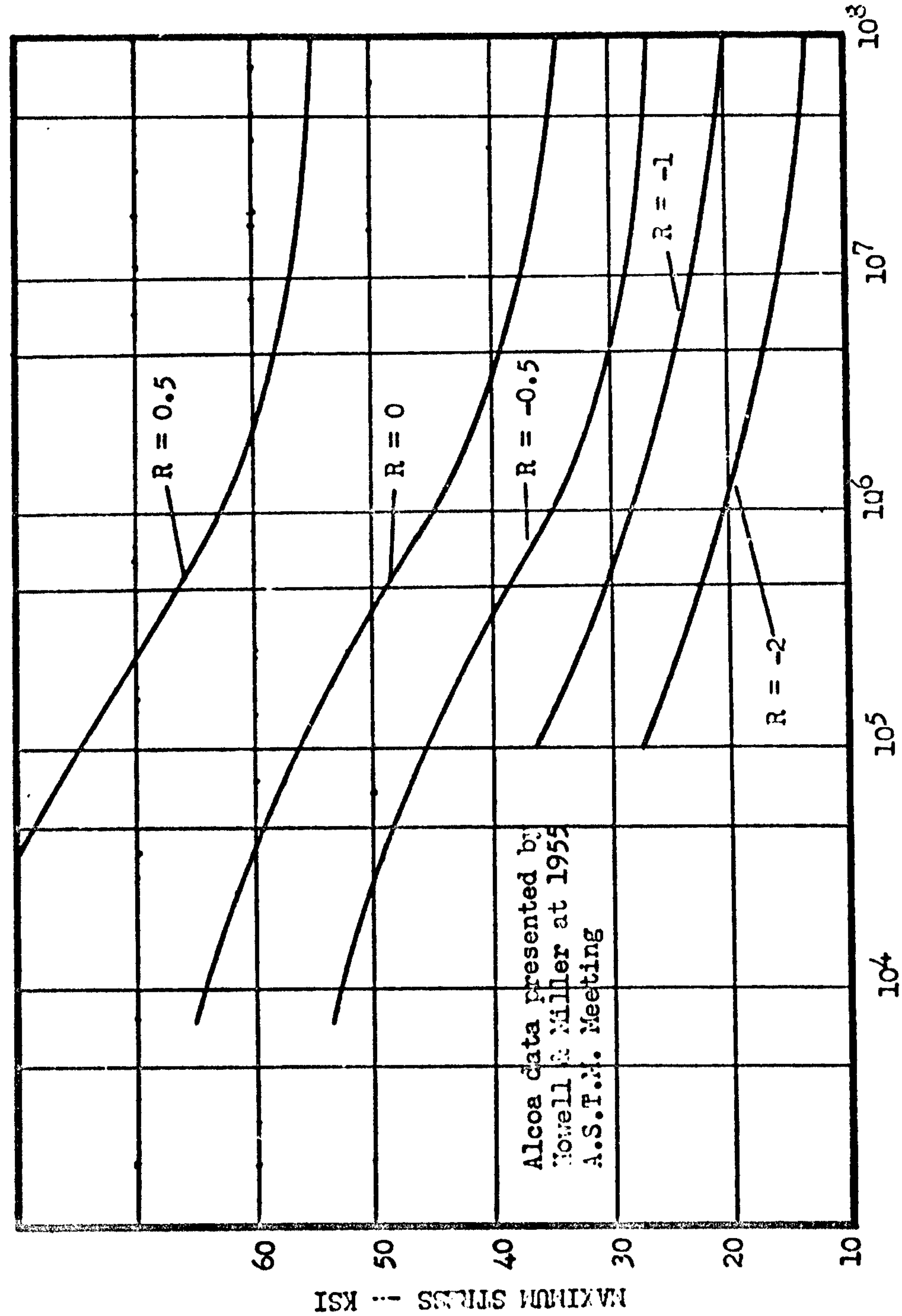


FIGURE 1. — DIRECT STRESS S-N FATIGUE CURVES
7075-T6 ALLOY

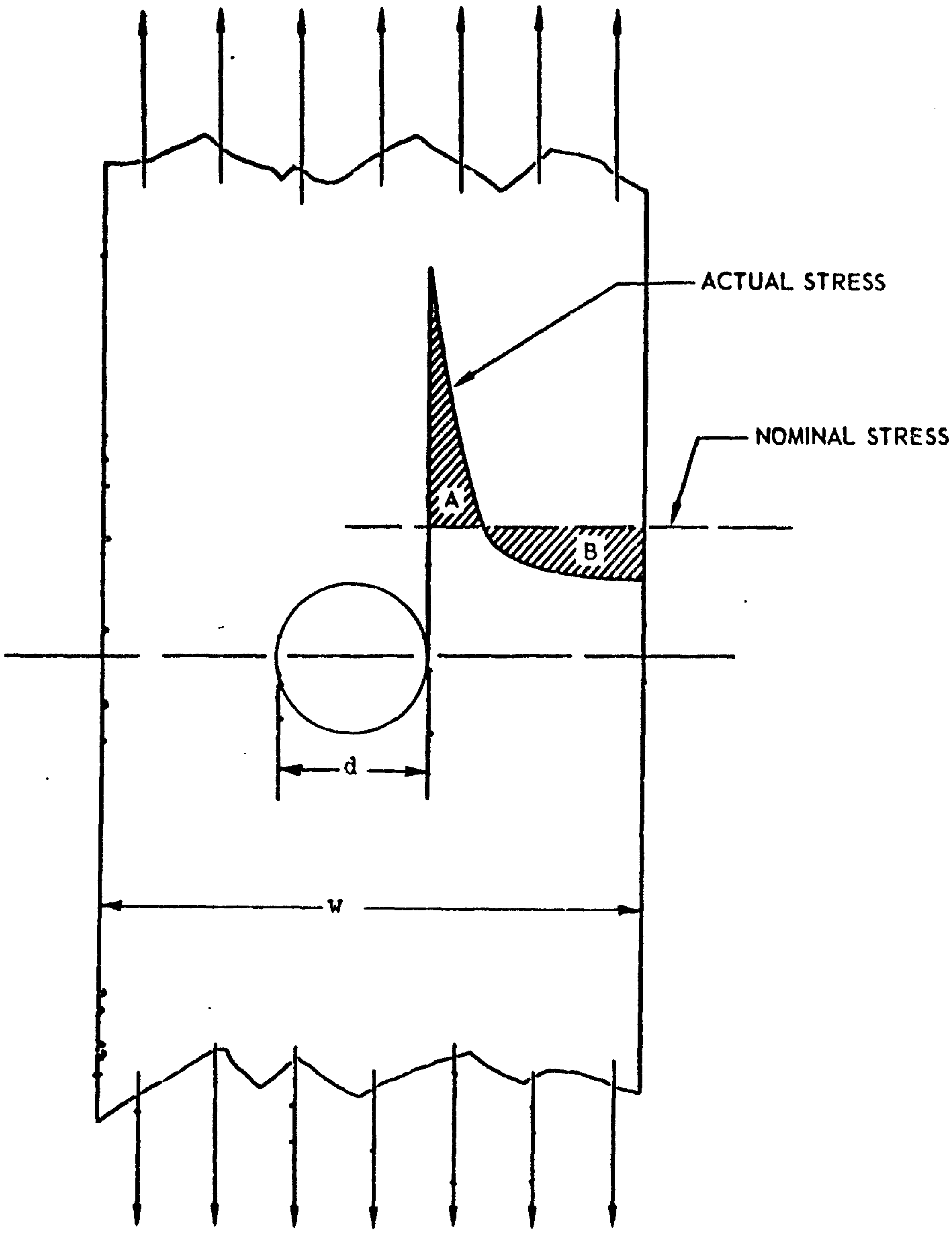


Figure 2 - - DISTRIBUTION OF STRESS ADJACENT TO HOLE

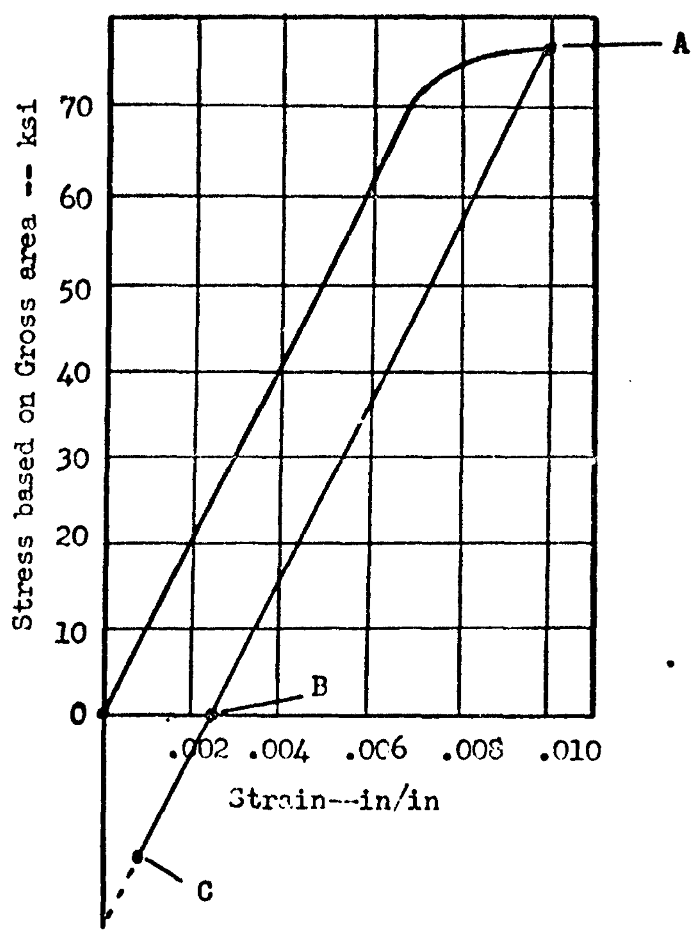


Figure 3 -- EFFECT OF PLASTIC FLOW ON STRESS CYCLE IN NOTCHED SPECIMEN

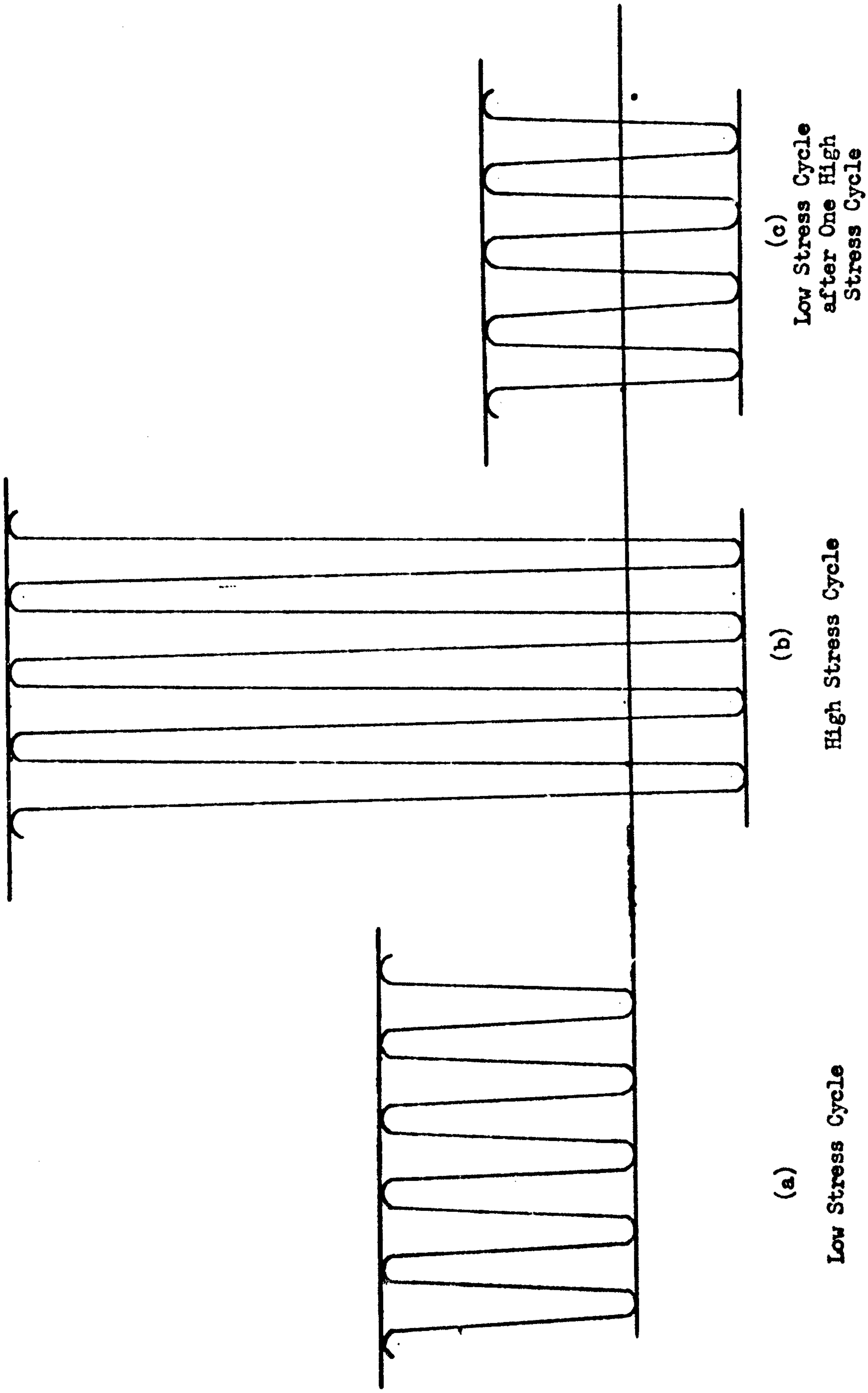


Figure 4. EFFECT OF HIGH LOADS ON SUBSEQUENT STRESS CYCLES

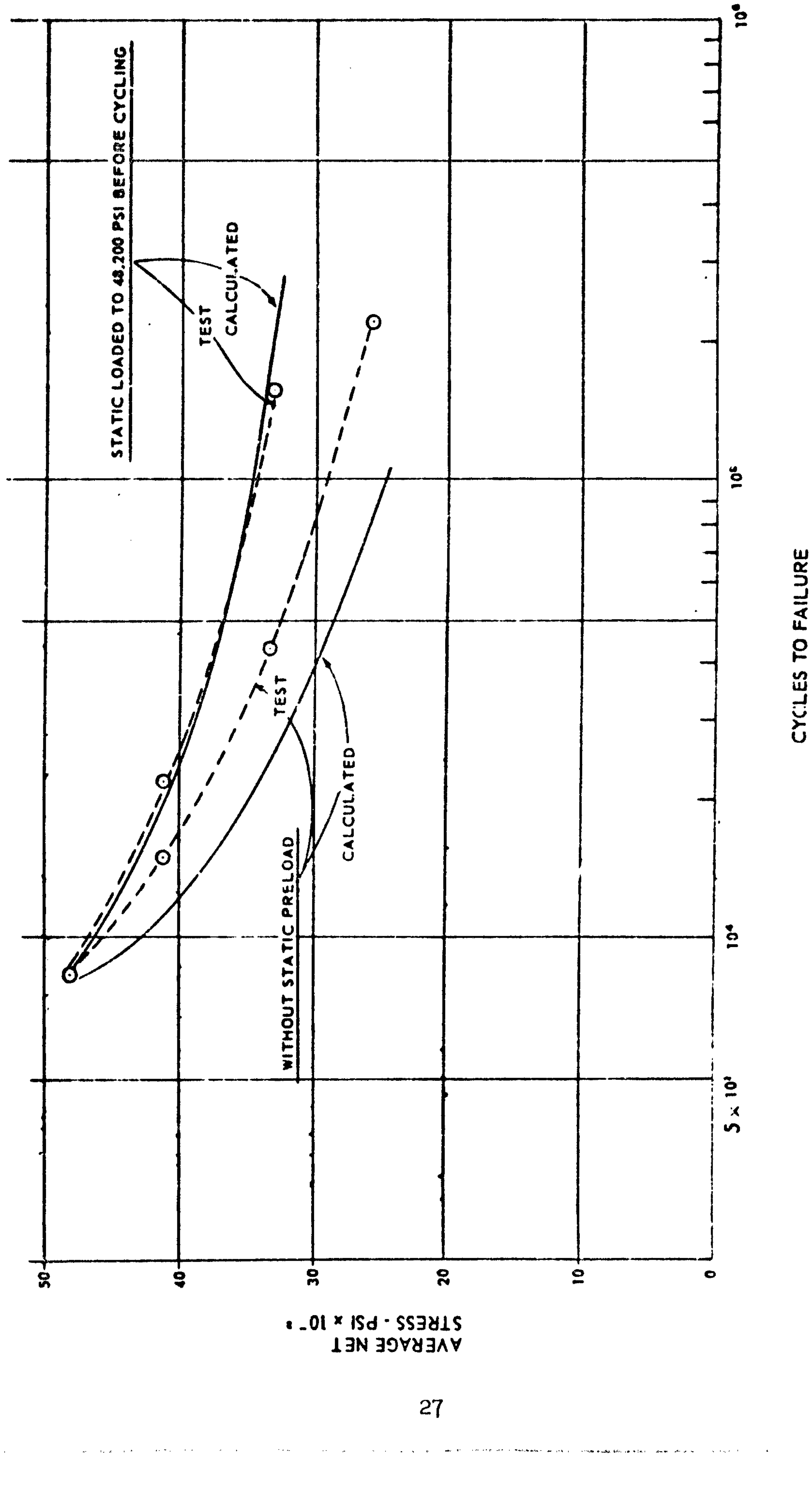


FIGURE 5 S-N CURVES FOR NOTCHED (K_t = 2.4)
7075 T6 SHEET

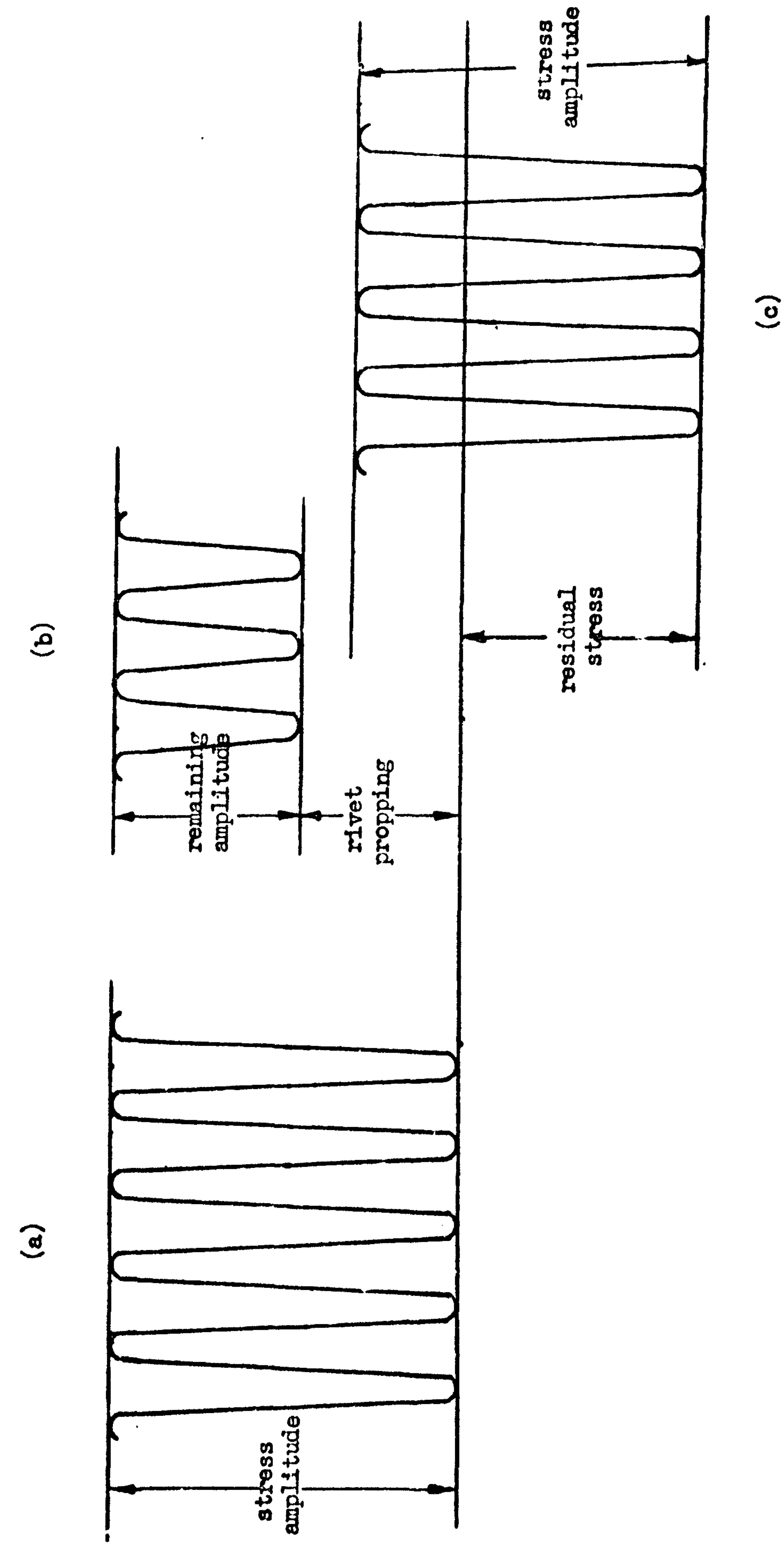


Figure 6 — EFFECT OF RESIDUAL STRESS ON AMPLITUDE

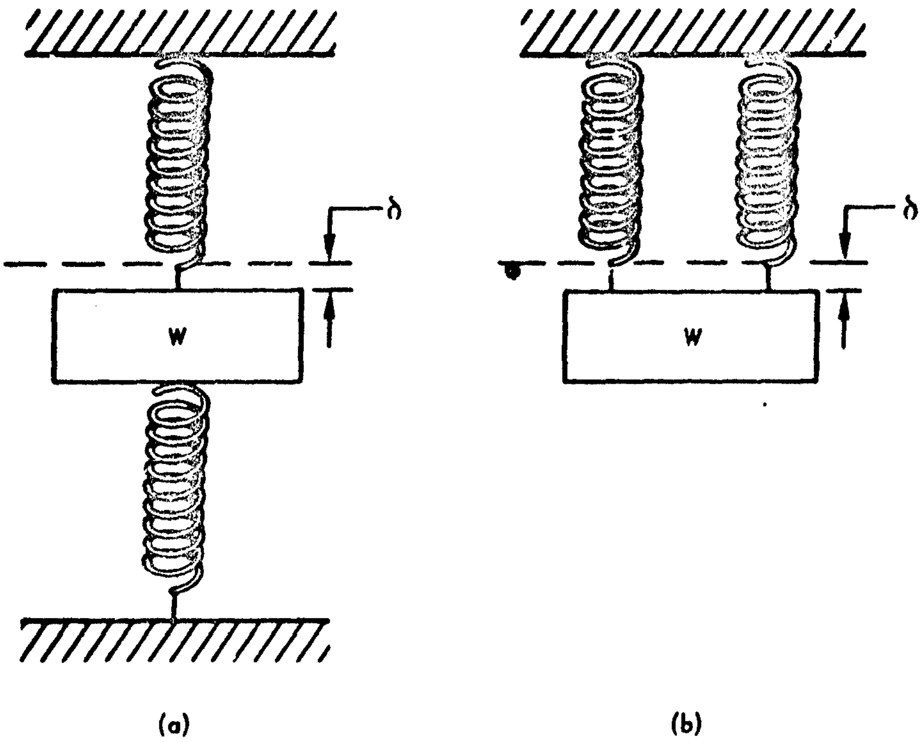
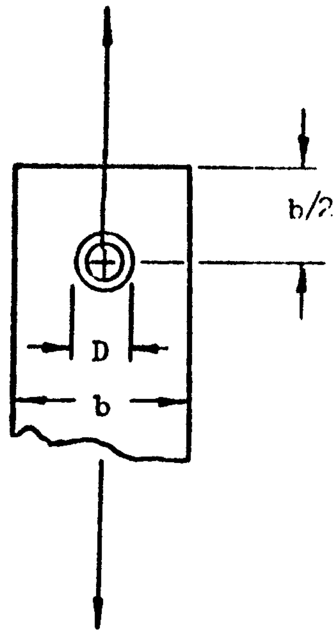


Figure 7. EFFECT OF SPRING COMBINATIONS



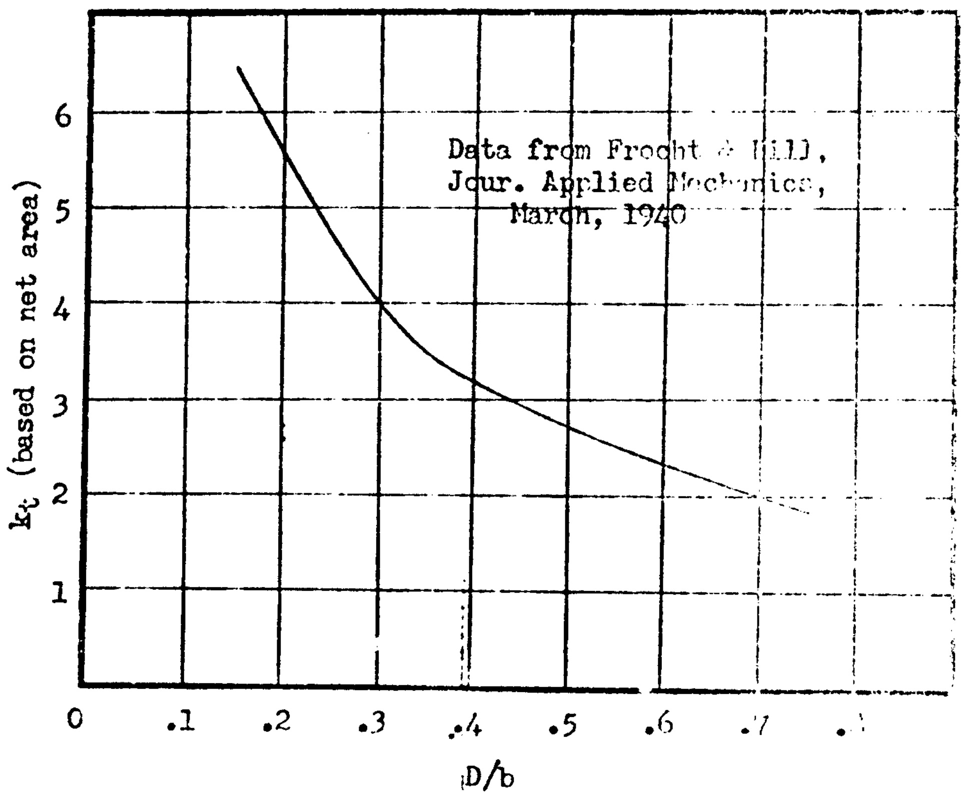


Figure 8—STRESS CONCENTRATION FACTOR FOR PIN LOADED LUG

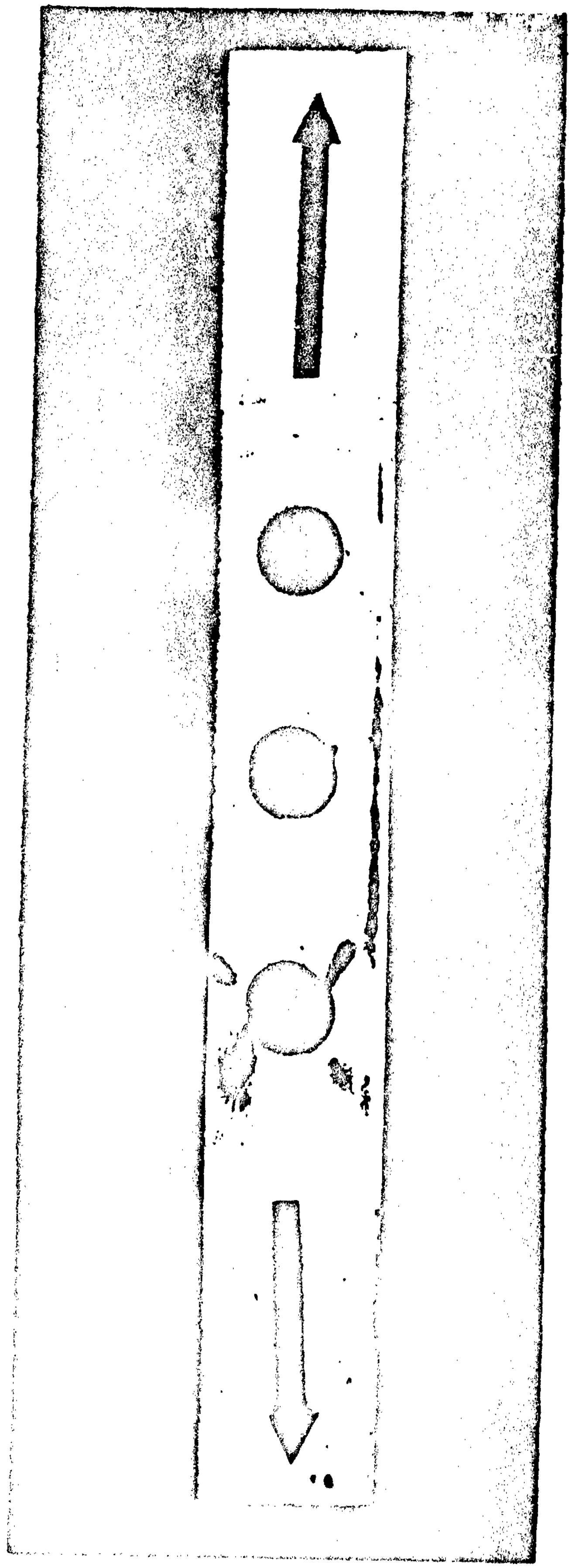


Figure 9 - - PHOTOELASTIC MODEL OF 3 RIVET LAP JOINT

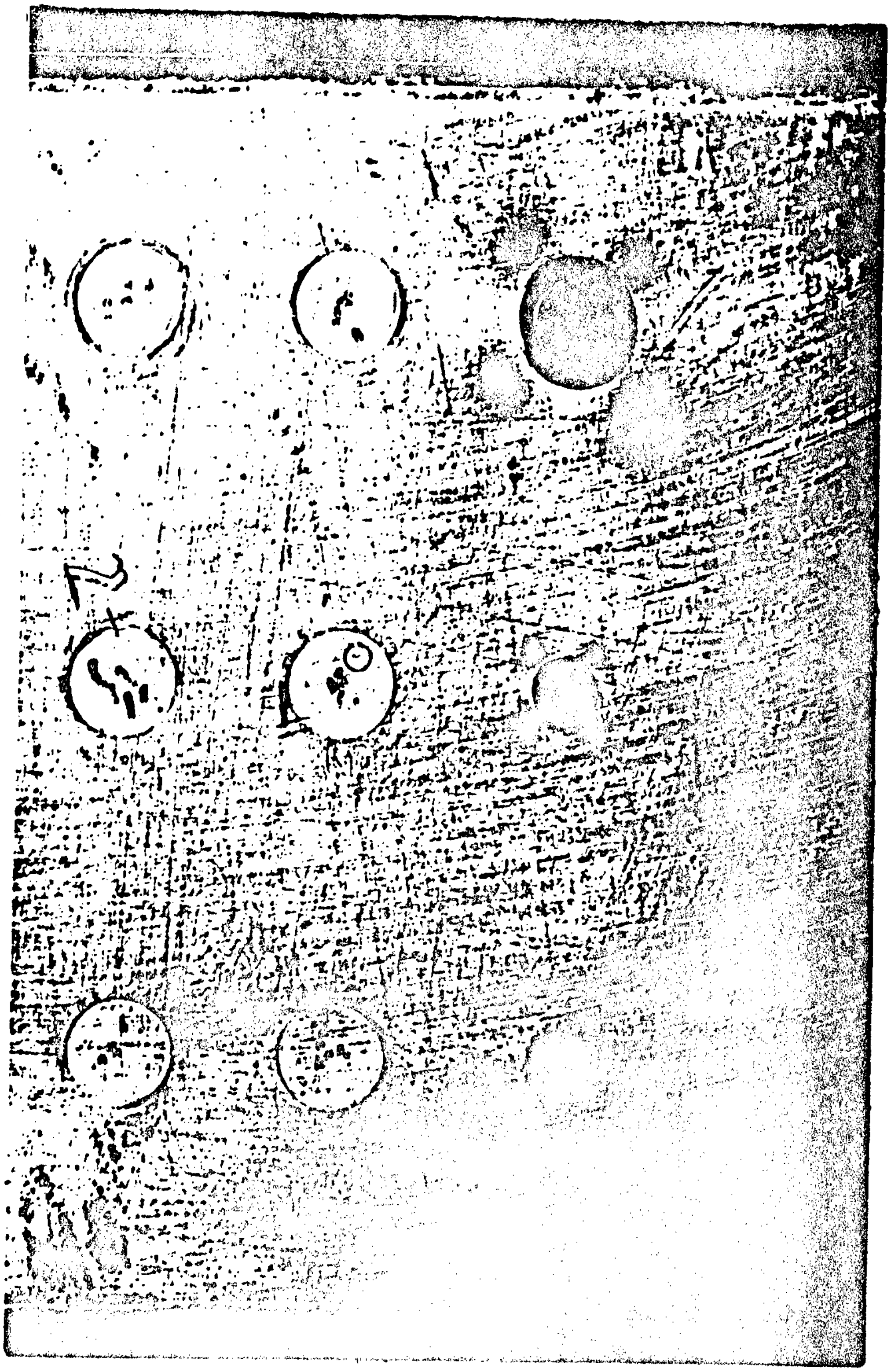


Figure 10 - - PHOTO-STRESS VIEW OF STRESS AROUND HOLES & RIVET

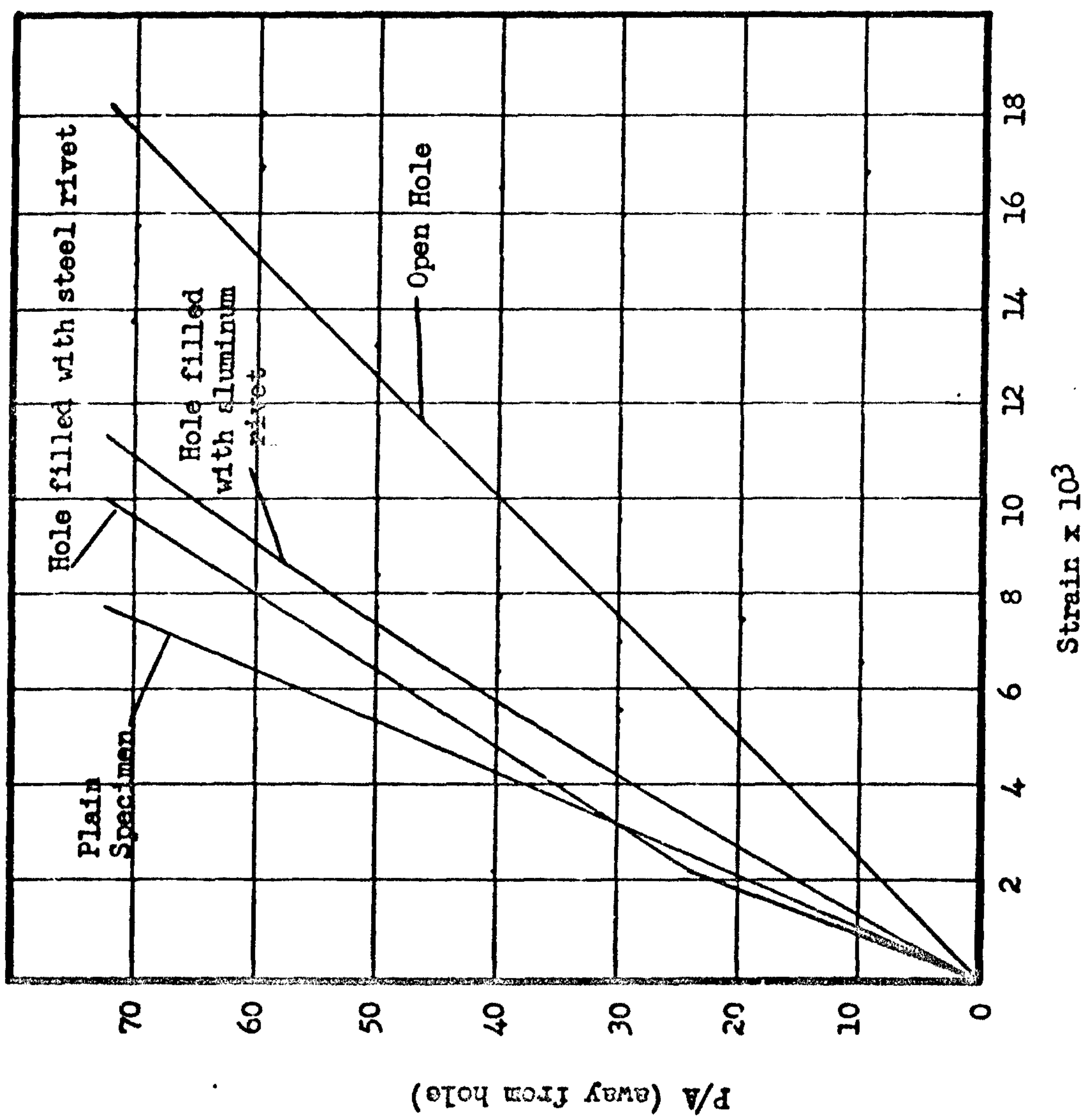


Figure 11 — PHOTOSTRESS SURVEY OF AXIALLY LOADED 7075-T6 SPECIMEN

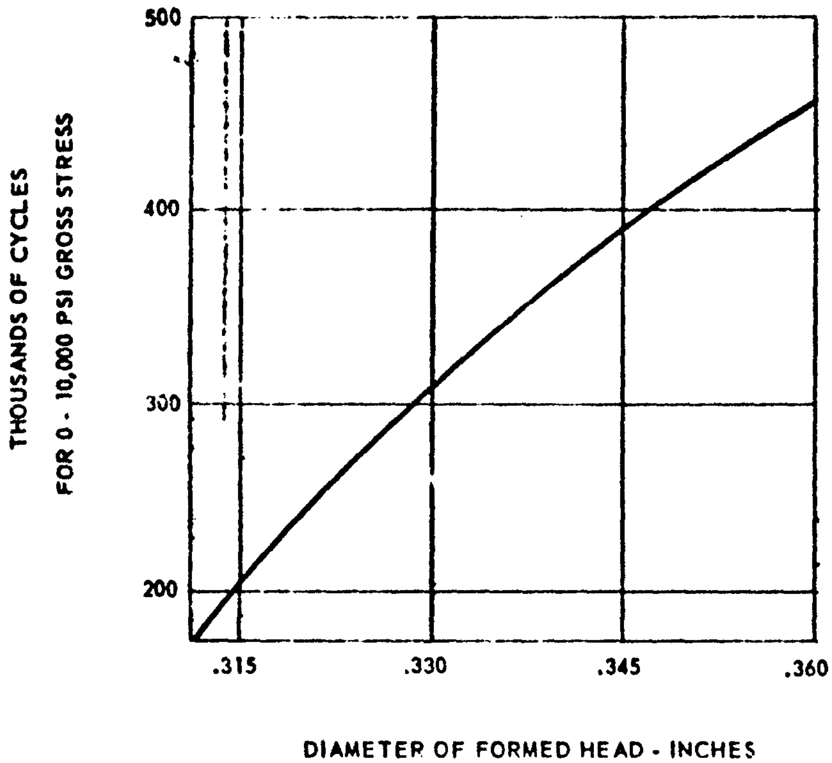


Figure 12. EFFECT OF RIVET DRIVING FORCE ON FATIGUE LIFE

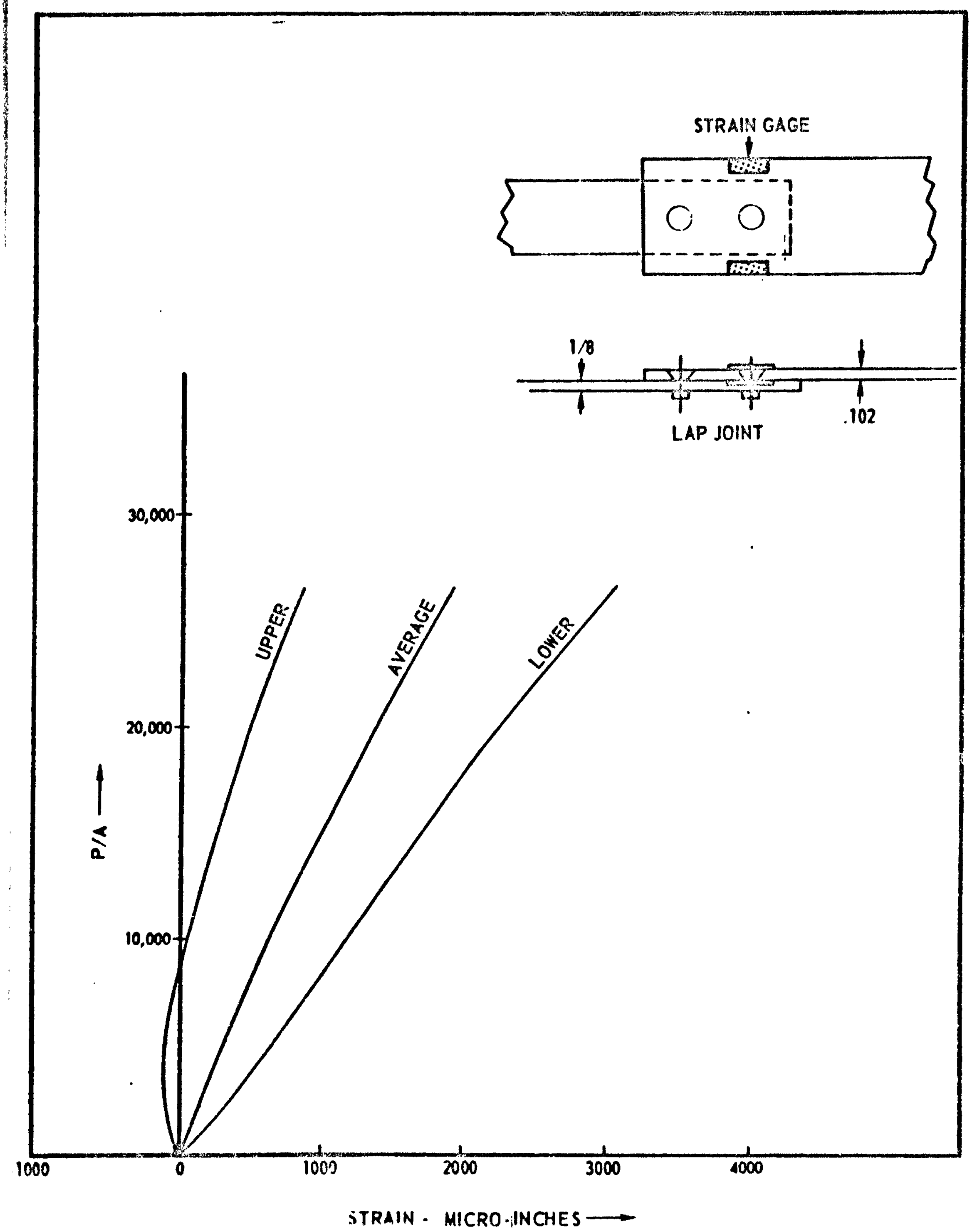


Figure 13 — STRAIN MEASUREMENTS ALONGSIDE RIVET IN LAP JOINT

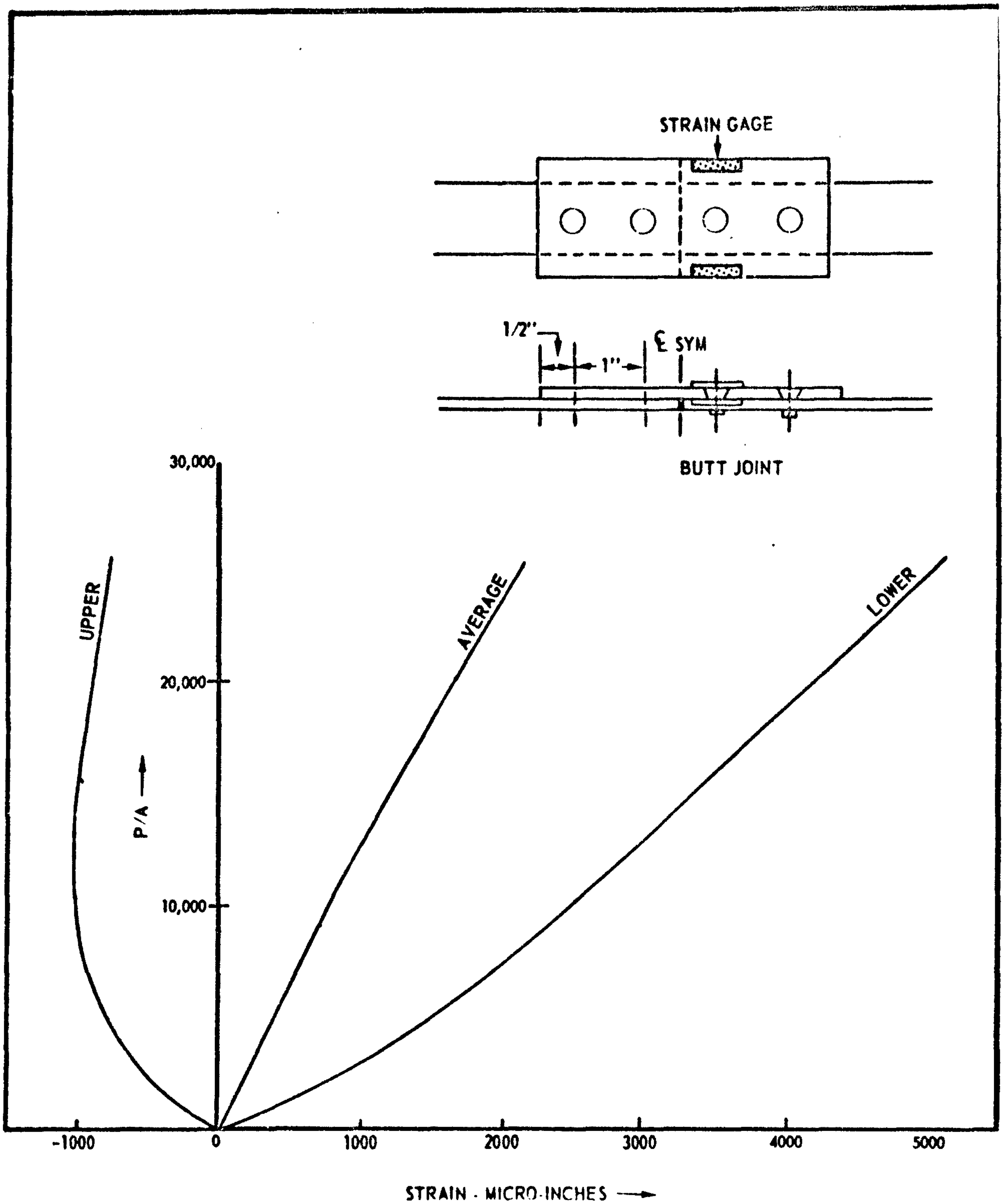


Figure 14. -- STRAIN MEASUREMENTS ALONGSIDE RIVET IN BUTT JOINT

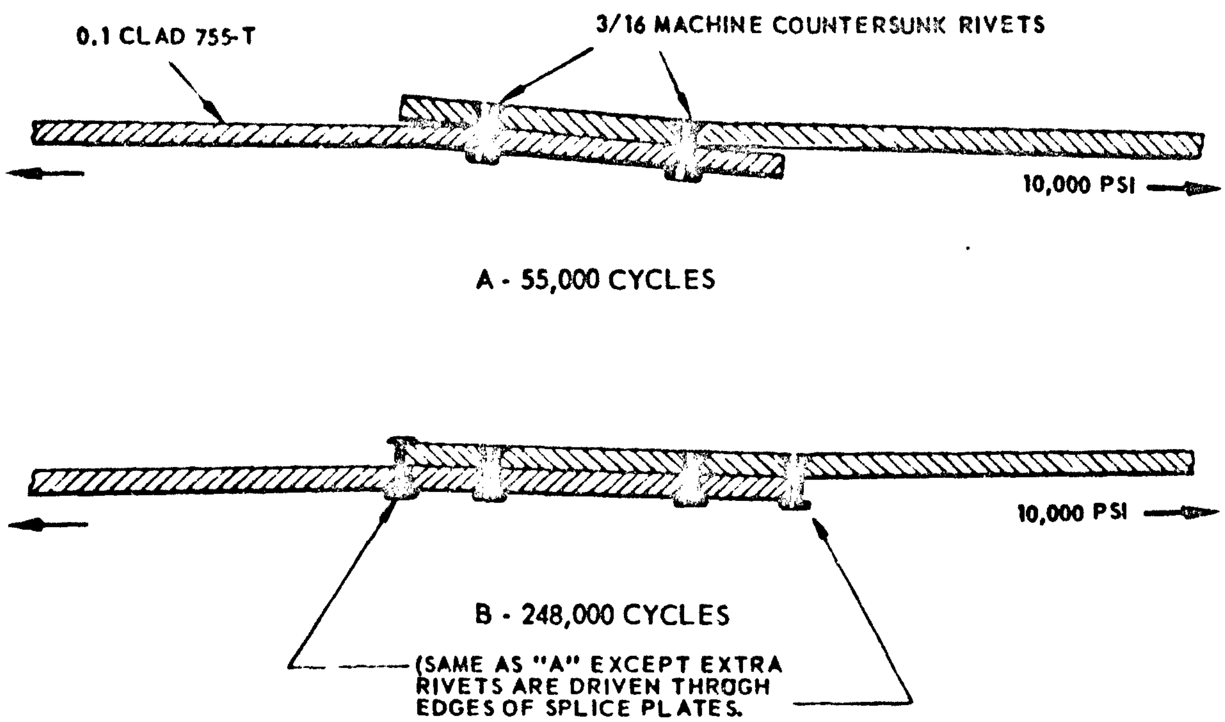


Figure 15 - - EFFECT OF EDGE DRIVEN RIVET ON FATIGUE LIFE OF LAP JOINT

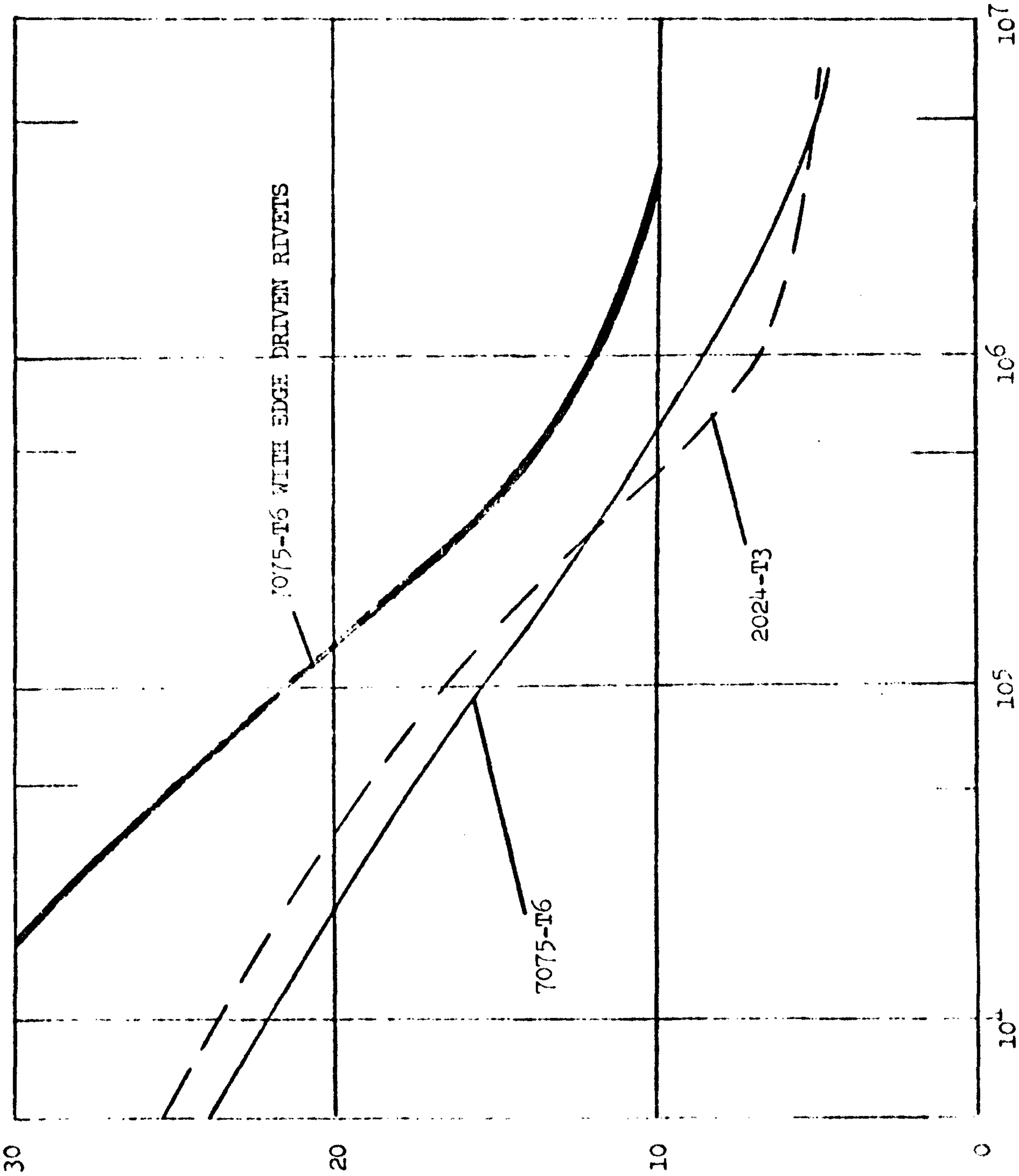
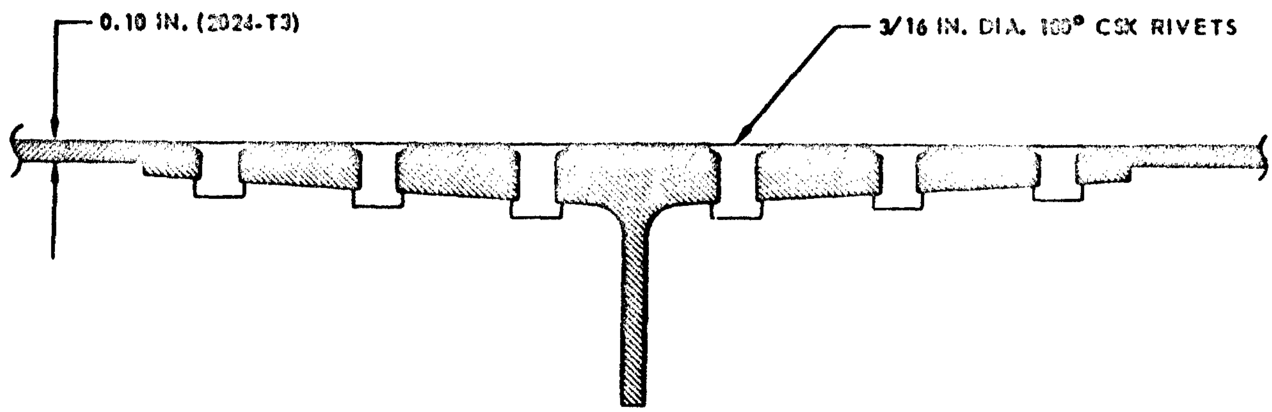
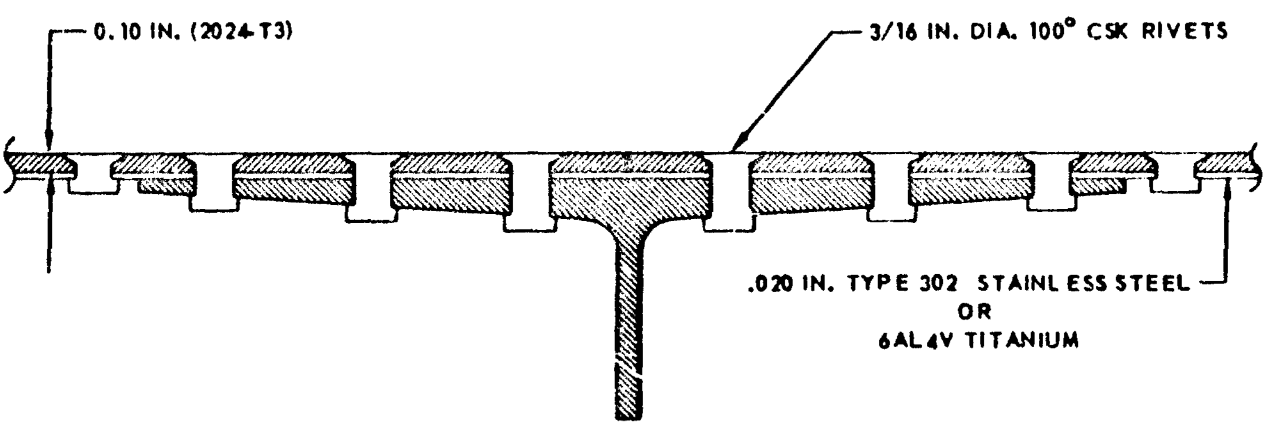


Figure 16 - - S-N CURVES FOR LAP JOINTS WITH EDGE DRIVEN RIVETS



(a)



(b)

Figure 17 - - TEST SPECIMENS WITH STEEL DOUBLERS

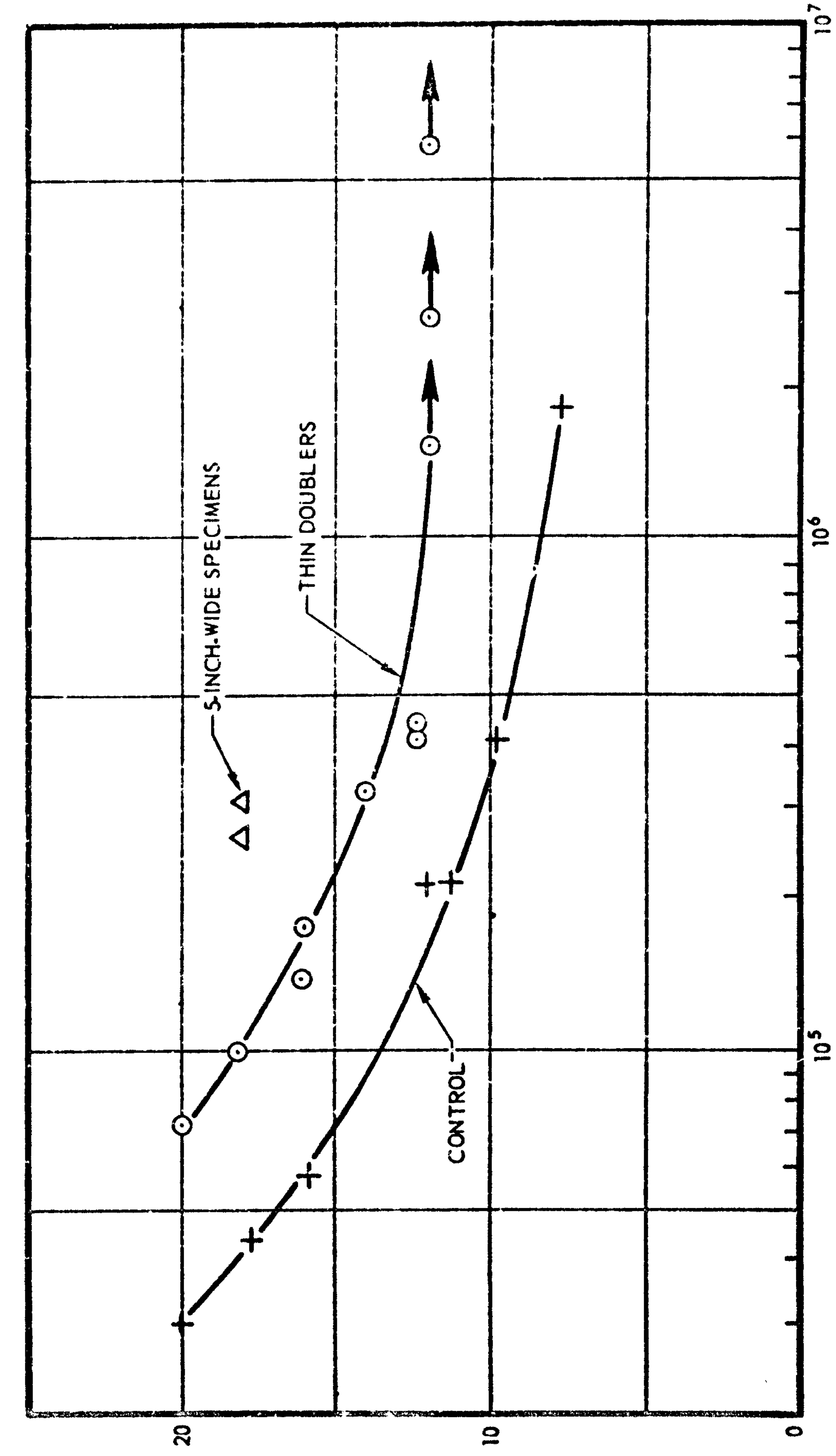
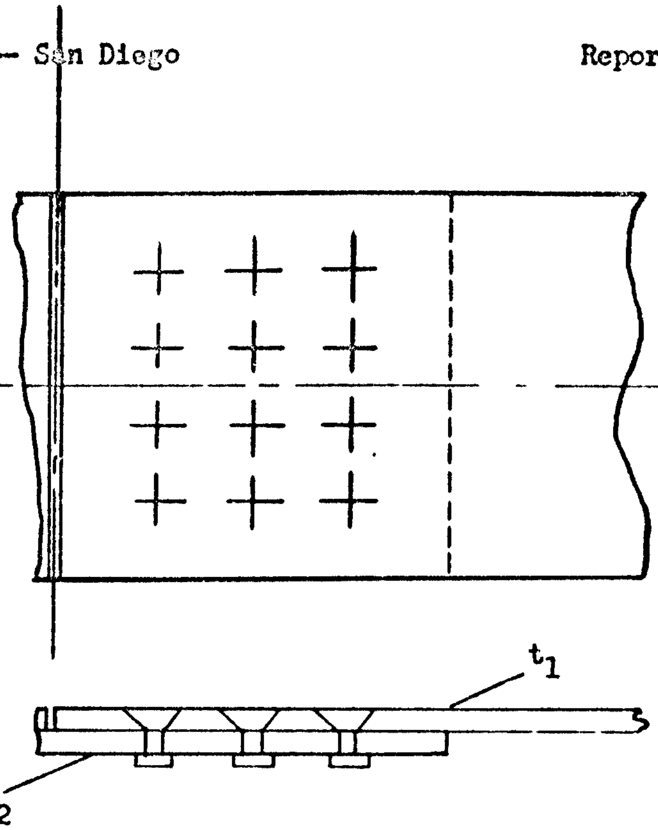
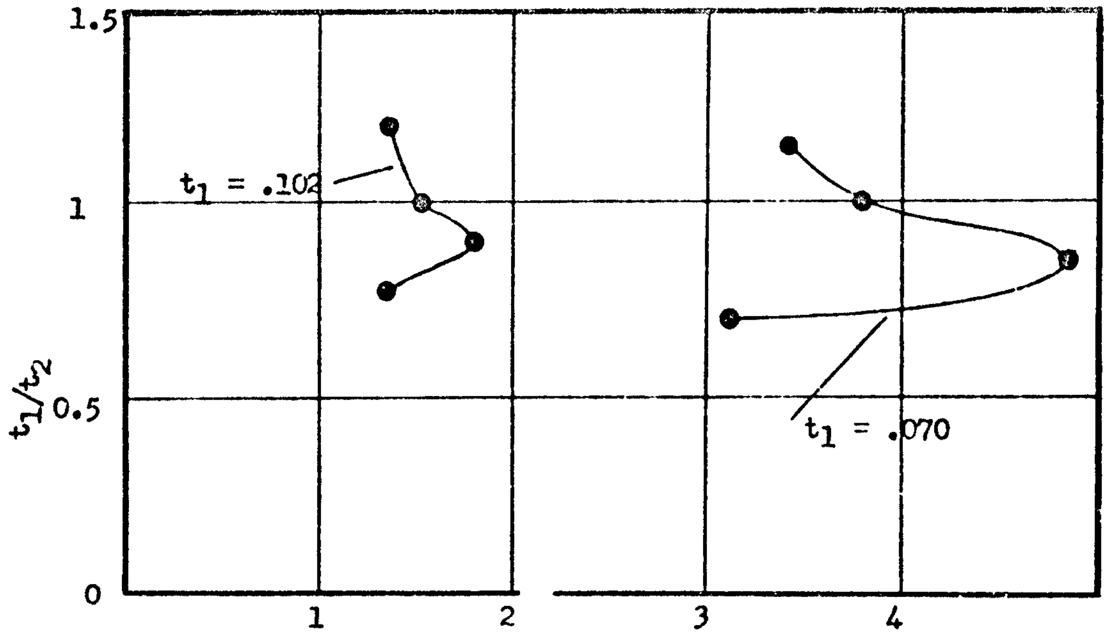


Figure 18 - - S-N CURVES FOR SPECIMENS WITH STEEL DOUBLERS





CYCLES TO FAILURE x 10^{-5}
For 10,000 psi gross stress -- R = 0.1

Figure 19 -- EFFECT OF DOUBLER THICKNESS ON FATIGUE LIFE

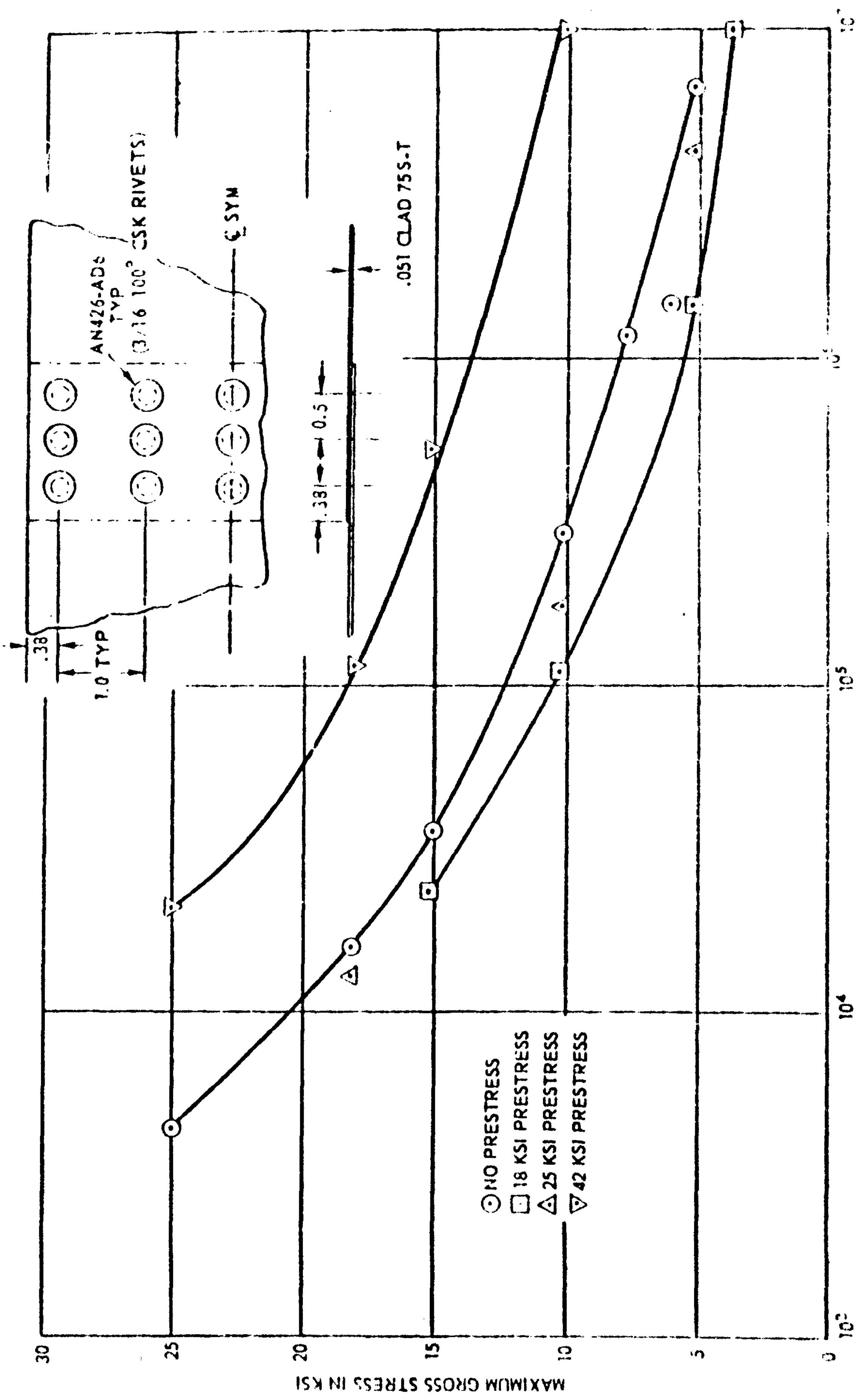


Figure 20 -- EFFECT OF PRESTRESS ON FATIGUE LIFE OF RIVETED JOINTS